

Experimental Study on a Stirling Cycle Machine of 100W Design Capacity

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Abstract: Environmental concerns are causing commonly used chlorofluorocarbon (CFC) refrigerants to be phased out of production. The less ozone-depleting HCFC's are regulating. The green house effecting HFC's are also likely to be regulated and banned in the next period. Accordingly, attention is drawn to the Stirling refrigerator, which is a perfect a freon free refrigerator. Moreover, The Stirling cycle has the highest theoretical cycle efficiency corresponding to the value of the Carnot cycle among the proposed thermodynamic cycles. The green house effect by carbon dioxide issue would make better recognizing the importance of efficient use of energy in terms of high energy conservation measures. The authors have designed and developed a 100 W class Stirling refrigerator for household use. And the prototype machine has been integrated with a 100 liter class refrigerator. The operating characteristics of this Stirling unit or the prototype machine have been evaluated. Moreover, the authors evaluated the machine driving engine mode using ultra-low temperature. As a result, the operational characteristics of the Stirling cycle machine have been clarified with respect to design factors. These results demonstrate that the Stirling cycle machine is one of the promising candidates as a new refrigeration system or a new generation system.

Keywords: Stirling cycle, Refrigerator, Cryogenic heat engine, COP

1 INTRODUCTION

A machine that utilizes the Stirling cycle could function as an engine that converts heat energy from an appropriate heat source into kinetic energy, or, by employing the reverse cycle, as a refrigerator that can achieve low temperatures or provide a heat-absorbing effect by the injection of kinetic energy from an electric motor. The Stirling engine has also been proposed as a driver for electricity generators and heat pumps, and its practical applications have been realized. Conventionally, fossil fuels and solar energy have been considered as potential heat sources; more recently, however, the practical applications of engines that utilize biomass or the waste heat generated from diesel engines as fuel have been pursued. Consequently, most Stirling engines operate in temperature ranges in which the temperature difference between the heat source and heat sink is between 100 K and 700 K, with the room temperature being at the lower end of the operating temperature range. However, information available on engines that utilize the room temperature as the heat source and the ultra-low temperature of liquid nitrogen as the heat sink is scarce. Engines that operate within such temperature ranges are called cryogenic heat engines. If their practical applications are realized, energy that has hitherto been wasted during the use of ultra-low-temperature media can be recovered in the form of electrical energy. On the other hand, the Stirling cycle refrigerator has been the subject of numerous researches and development activities, including the study of cryogenic

freezers as cold sources for infrared imaging devices[1]. Moreover, progress in its practical applications has also been reported[2]. However, reports on the study of freezers for household use are scarce. Household freezers differ from cryogenic freezers in that their operating temperature range lies between the temperatures that are suitable for refrigerating and freezing foodstuff up to the highest possible temperatures during a typical summer in Japan (i.e., between 233 K and 313 K); moreover, they must fulfill a predefined freezing capacity. In addition, from the perspective of energy conservation, a great emphasis is placed on the improvement of the coefficient of performance (COP), which represents the ratio between the cold energy that can actually be utilized and the input electrical energy. We previously developed a displacer-type 100-W Stirling refrigerator (hereafter referred to as the "prototype") and conducted experiments to evaluate its performance[3-4]. The results showed that the prototype could be adapted for household use. However, it also became clear that the mechanical loss in the prototype was greater than the indicated work. In this study, a detailed investigation of the losses was undertaken through experiments and analyses using the prototype. Further, the prototype was loaded onto a 100-liter class refrigerator and cooling experiments were performed in order to determine its refrigerating characteristics. Finally, the prototype was utilized as a cryogenic heat engine and the possibility of using it as a heat engine was empirically validated.

2 PROTOTYPE MACHINE

Table 1 lists the specifications of a prototype Stirling refrigerator connected to a freezer and evaluated. Since the prototype unit was designed for evaluation, a ball bearing was used for the mechanical section, and sliding parts were basically oil-free. Fig. 1 shows the structure of the prototype unit. The prototype unit minimizes the dead volume, and a regenerator is incorporated into the displacer. At the cylinder head and the tip of the displacer where the temperature becomes low, interdigitated fins are installed to promote the heat transfer between the cylinder wall and expansion gas.

Table 1 Specifications

Cooling capacity	100 W
Cooler wall temperature	233 K
Radiator wall temperature	303 K
Working fluid	Helium
Width×Height×Depth	260×320×130 (mm)
Bore×Stroke	60×20 (mm)
Mean pressure	0.7 MPa (Max.1.0 MPa)
Piston speed	16.7 Hz (Max.25 Hz)
Regenerator matrix	Wire mesh (Cu, #100)

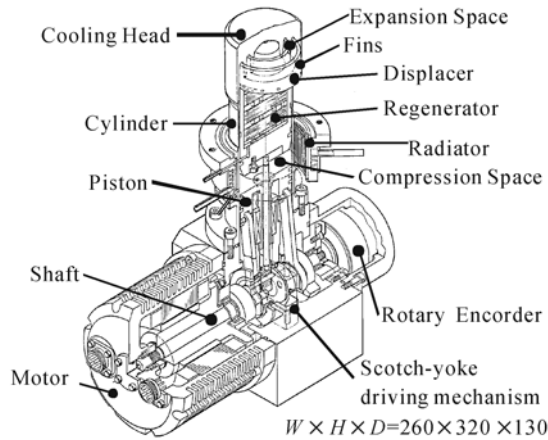


Fig.1 Schematic view of prototype refrigerator

3 EVALUATING LOSSES IN THE PROTOTYPE

3.1 Basic performance

The measured COP curves are shown in Fig.2 in order to evaluate the operating characteristics of the prototype refrigerator. When a cooling head temperature were 253 K and 233K, we obtained maximum COP of 1.05 and 0.70 respectively for the testing conditions of a 0.7 MPa mean pressure of Helium and a frequency of 11.7 Hz. Mechanical parts and performance of the electrical motor is not optimized, and major losses occur in power transmission process accordingly. However, it seems that these results demonstrate that the Stirling refrigerator will be a promising candidate for a household refrigerator.

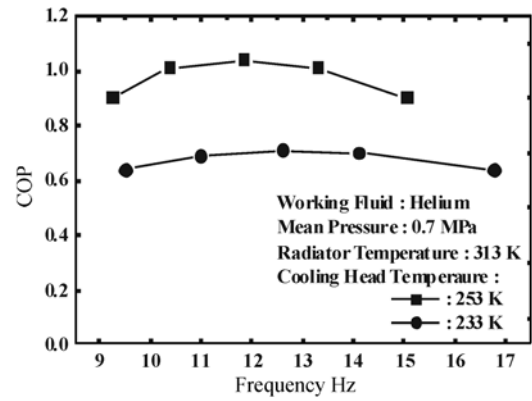


Fig.2 COP Curves

3.2 Motor loss

In order to evaluate the mechanical loss experimentally, the motor loss and indicated work must be subtracted from the input electric power; for this, the knowledge of the motor loss in the prototype is necessary. To determine the motor loss, the compression piston, displacer, and Scotch yoke mechanism were detached from the prototype and the input electricity was measured while the motor was running without any load on the shaft. Since an encoder was attached to the motor during this experiment, the friction loss in the encoder is considered to be a part of the motor loss. Therefore, the drive shaft was replaced with a single rod that was connected to the encoder. Ball bearings were used to support the drive shaft, and the friction loss in the ball bearings was assumed to be negligible. The results obtained from this experiment, which was conducted in order to measure the motor loss, are shown in Fig. 3. The effect of the changes in the charged pressure on the motor loss is almost negligible, and the loss increased almost linearly with the input frequency. The motor loss at a frequency of 0 Hz is equal to the inverter loss. In order to facilitate the handling of these experimental values in evaluating the mechanical loss, mentioned later in this paper, the motor loss W_m was approximated by using the method of least squares, as show in Eq. (1).

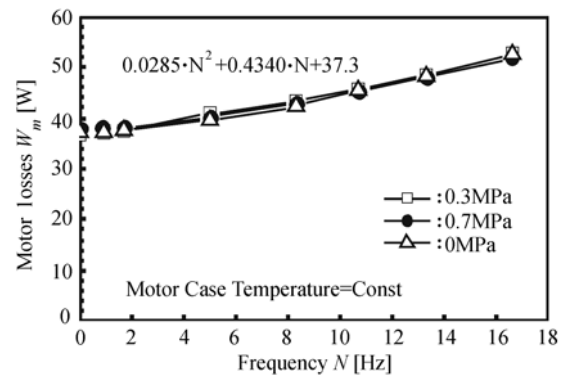


Fig.3 Motor losses

$$W_m = 0.0285 \cdot N^2 + 0.4340 \cdot N + 37.3 \quad (1)$$

3.3 Mechanical loss

The main sources of mechanical loss include the friction loss in the piston rings, crankshaft bearing, Scotch yoke shaft bearing, and conductive shaft bearing. Among these, the conductive shaft of the Scotch yoke is supported by ball bearings; since its frictional loss is smaller than the other losses, it is ignored. We empirically investigated the frictional losses in the piston ring and the Scotch yoke shaft bearing.

First, the friction loss in the piston rings on the compression piston was obtained by running the prototype with the displacer detached and by measuring the electrical input to the motor. Fig.4 shows the friction loss plotted against the charged pressure. The compression piston of the prototype was equipped with three tension rings. The measurement captured the sum of the friction losses in these piston rings. The friction loss increased with the charged pressure because the increase in the charged pressure caused the compression ratio to increase, leading to a greater perpendicular force of the piston rings against the cylinder wall. In particular, there was a rapid increase in the friction loss between 0.5 MPa and 0.7 MPa. From this result, we deduce that the relationship between the pressure of the piston rings against the cylinder wall and the pressure difference is not linear and that the perpendicular resistance is proportional to a value greater than the square of the pressure difference.

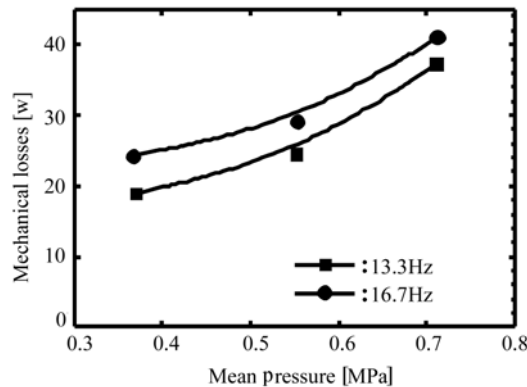


Fig.4 Mechanical losses of power piston

Next, the electrical input to the motor was measured while the prototype was run with only the compression piston detached. By subtracting the motor loss in the electrical input, we obtained the sum of the friction losses in the piston rings of the displacer, rider ring, and the bearing used in the Scotch yoke mechanism. Fig. 5 shows the sum of these losses plotted against the input frequency. This diagram clearly reveals that the rate of increase in the mechanical loss increases with the frequency. This tendency is mainly attributed to the increase in the inertial force of the displacer with the frequency, which in turn leads to an increase in the friction loss in the Scotch yoke shaft bearing. Moreover, it is noteworthy that there is almost no difference in the mechanical loss with the change in the charged pressure in the low-frequency region. The fact that a greater charged pressure leads to a greater mechanical loss

when the input frequency is higher is attributable to the increase in the friction loss in the piston rings as well as the effect of the flow loss in the working gas as it oscillates inside the regenerator matrix.

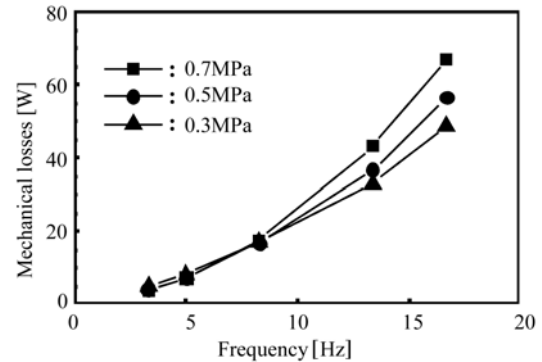


Fig.5 Mechanical losses as power piston less set-up

3.4 Heat conduction loss

One of the major losses that lead to the withdrawal of energy from the cold heat generated by the Stirling refrigerator is the heat conduction loss. In this section, the result obtained from an empirical study of the heat conduction loss is described. In the experiment, the cooling head was fitted inside a vacuum thermal insulation container and the cooling head wall was cooled to below 173 K. The prototype was then stopped, and the change in the cooling head wall temperature was monitored over time. The heat conduction loss was computed by multiplying the change with half the heat capacity of the cylinder head, displacer head, and regenerator. In addition, the loss due to radiation and the heat generated by the piston rings due to friction were not considered since these losses are deemed negligible. Fig. 6 shows the heat conduction loss, Q_{loss} , plotted against the temperature difference between the radiator and the cooling head wall. As seen from the graph, there is a rapid increase in the loss that appears to be the effect of radiation in the region where the temperature difference is large. However, we observe a linear increase in the loss in temperature ranges that are typical for household refrigerator use. The least square approximation of the plotted curve is given in Eq. (2).

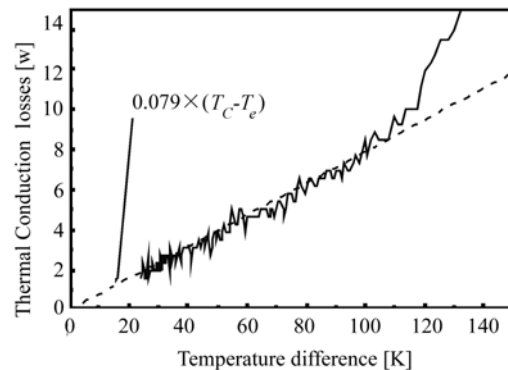


Fig.6 Thermal conduction losses

$$Q_{\text{loss}} = 0.078 \times (T_c - T_e) \quad (2)$$

Q_{loss}	Thermal conduction losses	W
T_c	Compression side temperature	K
T_e	Expansion side temperature	K

4 COOLING EXPERIMENT OF THE REFRIGERATOR

We evaluated the performance of the prototype machine by attaching it to a household freezer. Fig.7 shows a secondary-refrigerant direct cooling system. As the secondary refrigerant, an ethylene glycol solution (hereafter referred to as brine) was used. Box-shaped copper plates were attached to a heat exchanger placed in the chamber to ensure a large area for heat transfer and to reduce the flow loss of the brine. A 1/4-inch copper pipe was fitted around the copper plates and

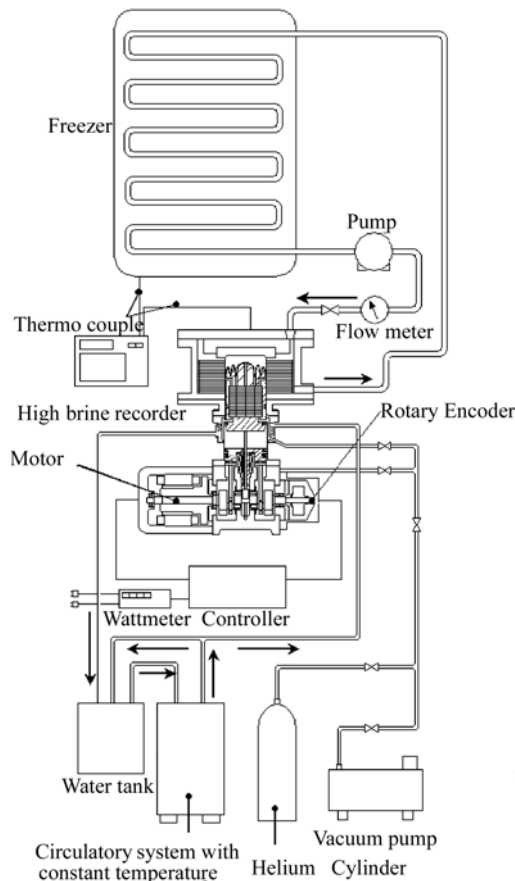


Fig.7 Secondary-refrigerant direct cooling system

brazed. The secondary refrigerant transfers heat removed from the low-temperature-side heat exchanger installed at the cooling head of the prototype machine to the chamber. After cooling the chamber using the direct-cooling heat exchanger installed in it, the heat removed returns again to the low-temperature-side heat exchanger of the prototype unit via a pump and a flowmeter installed outside the chamber. In the high-temperature-side heat exchanger of the prototype unit, a flow of cooling water is continuously supplied by a constant-temperature circulator at a constant temperature of 315 K and a constant flow volume. Prior to the initiation of the

cooling experiments, the experimental apparatus including the freezer chamber was left open to the atmosphere for a sufficient period. The inlet pressure of the prototype machine was set at 0.7 MPa, and the number of rotations at 16.7 Hz at the start of the operation and changes in temperature at each part were measured. Fig.8 shows changes in temperature at each part of the freezer. The temperature at the wall of the cylinder head rapidly and immediately decreased after operation began, and the cooling rate was high.

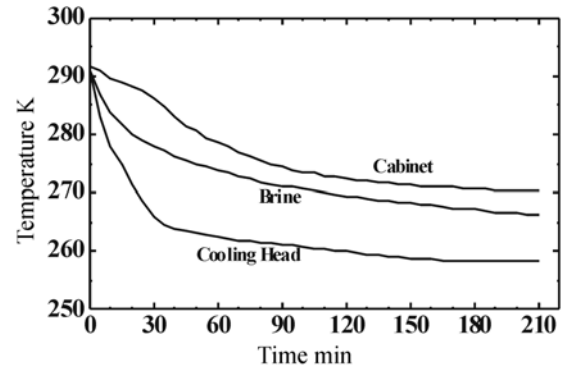


Fig.8 Pull down test results

The cooling rate then became moderate 40 min later. In contrast, changes in the temperature of the brine and inside the chamber were moderate from the beginning, which is different from the temperature change at the wall of the cylinder head, due to the thermal characteristics of the brine; the selection of a secondary refrigerant should be reconsidered in the future. The temperature inside the chamber was 270K 210 min after the start. Additional future improvements are anticipated.

5 CRYOGENIC HEAT ENGINE MODE

Using liquid nitrogen as the cooling medium in the radiator on the compression side, we tested the operation of the prototype as an engine by using the difference between the temperature of liquid nitrogen and the room temperature. In comparison to the refrigeration mode, the rotation was reversed: the expansion space in the refrigeration mode (i.e., the cylinder head side) became the compression chamber and the compression space became the expansion space. The liquid nitrogen was injected into the jacket cooler that was used with the cylinder head for transporting cold heat in the refrigeration mode. Ribbon-shaped electric heaters were wrapped around the radiator in which cooled water was circulated in the refrigeration mode, and the radiator was thermally insulated. The DC servomotor that was used as the mechanical power source in the refrigeration mode was removed and the DC motor that served as the starter and electricity generator was connected to the device via a flexible coupling. The electricity generated was measured by using this set-up. Fig.9 shows the pressure-volume (PV) curve in the case where the charged pressure is set to 0.8 MPa. The indicated expansion and compression work are 194.6 W and 148.6 W, respectively, and as a result, the indicated work is 46 W. From this result, it is evident that the prototype can operate as a cryogenic heat

engine. Fig. 10 shows a simplified energy balance of the cryogenic heat engine. The mechanical losses in the Scotch yoke and the piston ring were roughly estimated based on the measurement from the loss analysis of the refrigerator. There are two reasons for the small value of the indicated work. The first is that the prototype design, with regard to its operation volume and sealed structure, was optimized for use in temperatures ranges suitable for household use. The second reason is that the volume ratio of the expansion and compression spaces was not optimized for the present experiment, which was conducted in temperatures between 77.2 K and 286.7 K. In addition, the sealing on the O-ring became less airtight in extremely low temperatures, causing the working gas to leak. Thus, a refinement in the seal structure and an optimal volume ratio for a given temperature ratio of the heat source to the heat sink are required for cryogenic heat engine design. Only a small amount of electricity was generated due to the low efficiency of the DC motor and since the mechanical losses in the flexible coupling and the brush inside the motor were large. In addition, there was a large amount of heat input beside that generated by the electric heater, and the development of a regenerator and a heat exchanger on the high temperature side that have been optimized for cryogenic heat engine is required.

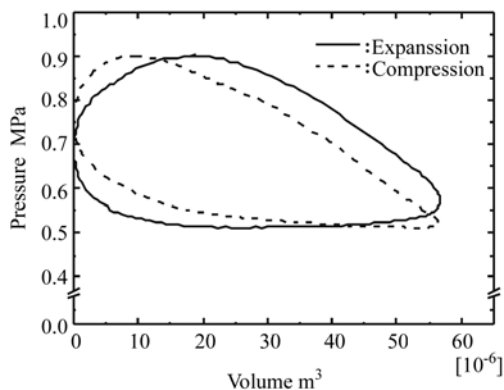


Fig.9 PV-Diagrams

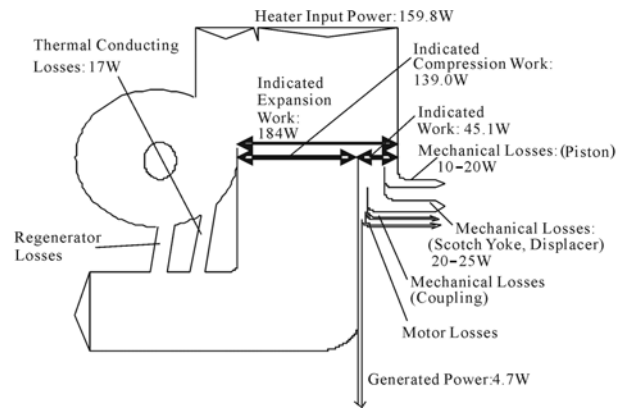


Fig.10 Heat balance

6 CONCLUSION

We have carried out a loss analysis of a Stirling refrigerator that operates in a temperature region that is suitable for household use; studies for the development of such a refrigerator are virtually nonexistent. The prototype was utilized as a cryogenic heat engine that operates between the temperatures of liquid nitrogen and room temperature and it was empirically studied. We obtained the following insights from these experiments:

- (1) The friction loss in the shaft bearing on the Scotch yoke is significant.
- (2) The heat conduction loss was studied and a correlation equation was derived.
- (3) It was demonstrated that the prototype could operate as a cryogenic heat engine, but various parameters, including the volume ratio and the seal structure, require optimization.

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